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A Study on a Combined Light Pipe and Solar-heated Ventilation Stack

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Abstract

A theoretical and experimental study has been made of the performance of a vertical light pipe of height 3 m and cross-section area 0.0625 m^2 surrounded by an air duct of total cross-section area 0.23 m^2 above a room of height 3.8 m and floor area 9 m^2. Heat transfer from the hot water jacket to the air duct is assisted by stainless steel fins. The ventilation of the room, due partly to the buoyancy of the air in the duct and partly to the wind effect, amounted to nearly 10 air changes per hour. The light pipe has specular reflecting walls. It was found that the transmission of daylight through the light pipe in the middle of a partly cloudy day varied in the range 56-84%.

Keywords: Natural ventilation, Daylighting, Thermal buoyancy, Light pipe, Counterflow heat exchanger, Solar water heater.

1. Introduction

Natural ventilation and daylighting are two methods of reducing energy needs in buildings. Normally, they are separate and independent operations. However, Oakley et al. [1] have put them together in a combined light pipe and ventilation stack. We have taken up this idea and investigated the performance of a similar stack surrounded by a solar-heated water jacket to enhance the natural ventilation for application in buildings in a tropical climate.

Air flow by natural means is due to pressure differences caused by wind and thermal buoyancy. Control of wind speed and direction is difficult whereas natural ventilation by the stack effect can be achieved by suitable design. Recent research has shown that passive night ventilation using the cool ambient air can significantly reduce the cooling load in air conditioned buildings with a massive envelope and increase thermal comfort levels in buildings without air conditioning [2].

Solar energy has been used for inducing natural ventilation for centuries [3], and demonstrated uses of solar energy for ventilation are numerous worldwide. The driving force in a solar chimney is the buoyancy produced by solar heating. Air from the ventilated space enters the chimney inlet and fresh outdoor air enters the space through open windows and doors [3]. Another popular technology in buildings is the solar water heater with circulation by thermosyphoning. Circulation is continuous as long as the sun is shining [4]. Solar heated water that has accumulated in a storage tank during the day can be pumped through a water jacket around the ventilation stack to enhance the buoyancy at anytime.

Light pipes are a simple means of directing daylight into interior spaces [5]. Using light pipes, the daytime energy consumption for artificial lighting can be significantly reduced. Our research is concerned with a solar heated ventilation stack for residential buildings in which the stack also functions as a light pipe during the daytime. The basic light pipe design considered for this study has a passive light collector (without sun tracking) on the roof of a building.

The two functions of our stack, namely ventilation and daylighting, have been studied separately by experimental measurements and mathematical modeling. The experiment was conducted on the Bang Khun Tien campus of King Mongkut's University of Technology Thonburi 25 km southwest of Bangkok.

2. Experimental Set Up

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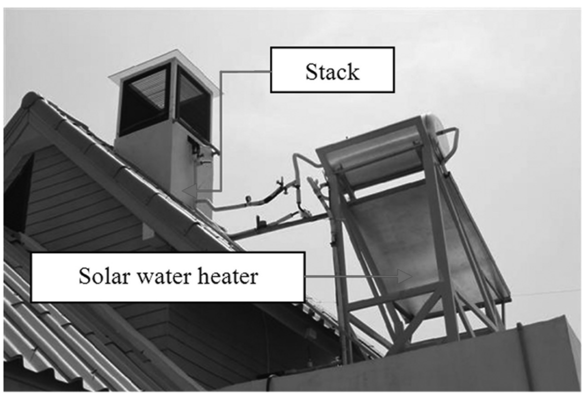

Figure. 1. Light pipe and Solar heated ventilation stack.

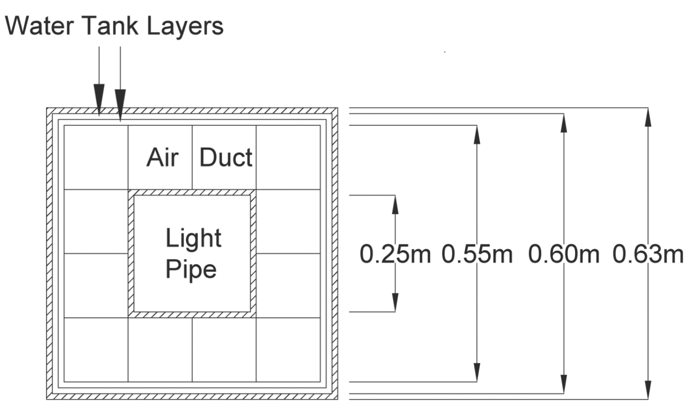

Figure. 2. Horizontal cross-section of stack. Light pipe has specular reflecting inner surface. Air duct is divided into twelve square channels by stainless steel fins. Water jacket is divided into an outer static layer and an inner circulation layer. Layers of thermal insulation are shaded.

The configuration of the light pipe and solar-heated ventilation stack is shown in Fig. 1, and a cross-section of the stack interior, which has five concentric layers is shown in Fig. 2. The fins, which transfer heat from the water jacket to the air in the duct, are of stainless steel 1.2 mm thick. It was estimated initially that the heat transfer rate between the water and the air would be about 50 WK^{-1}. The insulation layers are of fiberglass 12.5 mm thick.

Figure 3 gives a vertical side view of the stack, which is 3 m high. Thermally insulated half-inch copper pipes were used between the solar water heater and the stack water jacket. Hot water from the solar collector tank was pumped downwards through the water jacket causing the air inside the stack to rise in counterflow.

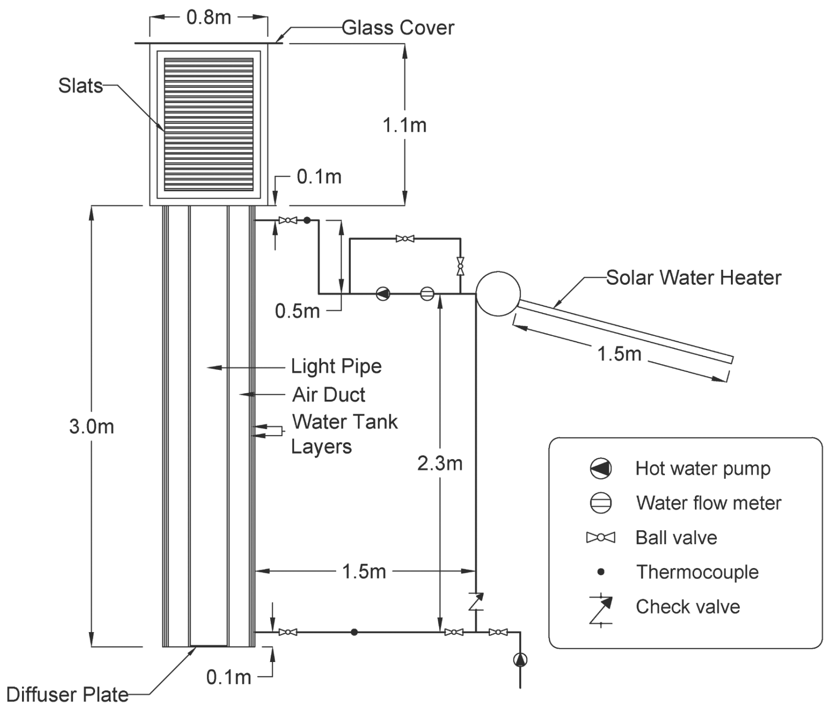

Figure. 3. The three system components.

The light pipe transmits daylight to the interior space, which has four closed doors, as shown in Fig. 4. A chamber of 1.1 m high at the top of the stack has downward-pointing black-painted slats to allow air to escape and keep out rain. The glass cover at the top has transmittance 88%. The walls of the light pipe are covered with specularly reflecting plastic film having reflectance 99%. The bottom of the light pipe is covered with a transparent plastic plate having concentric circular grooves to diffuse the light.

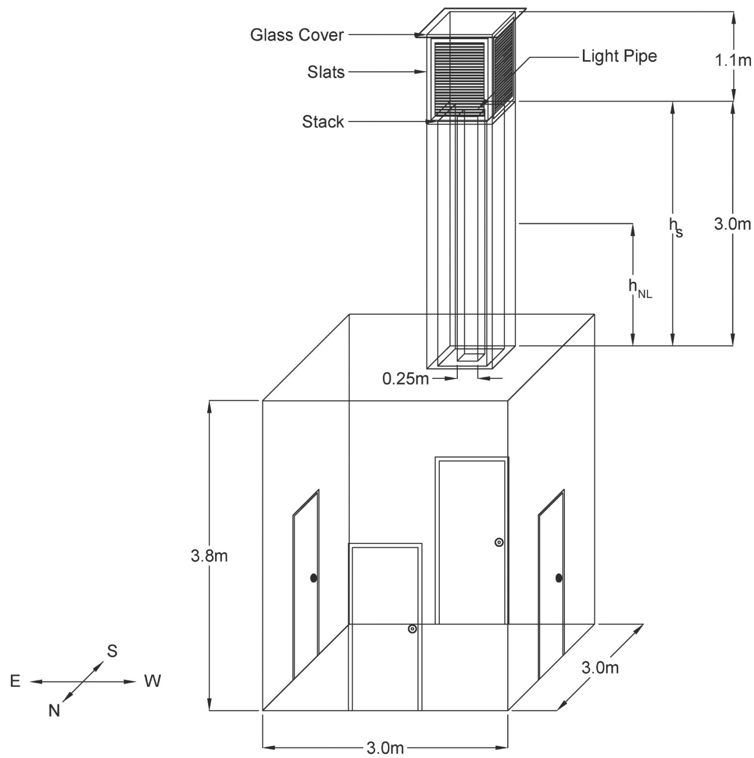

Figure. 4. The experimental room and stack (sloping roof not shown).

This set up is different from that of Oakley et al. [1] in that we have used a longer light pipe (3 m instead of 2.2 m) with a cross-sectional area of 0.0625 m^2 instead of 0.0363 m^2, and a full size room (34.2 m^3) at a temperature over 30 °C in a tropical climate in June instead of a small box (2.2 m^3) with a controlled temperature of 20 °C in January in England. Thus, although the concepts in the two cases are the same, the daylight and thermal conditions are quite different.

3. Mathematical Model

3.1 Volumetric rate of air flow through the stack

The airflow through the stack is determined by the following factors [6]:

(1) The buoyancy of the heated air in the stack, (2) the wind effect on cracks around the doors, (3) the wind effect on the top of the stack, and (4) the friction loss in the stack.

We assume there is a neutral pressure level where the pressure in the stack equals the ambient air pressure P_a at the same level outside the stack at height h_{NL} above the bottom of the stack (see Fig. 4). We also assume, as an approximation, that estimates of the airflow through the stack can be made using a single effective stack air temperature T_s between the inlet temperature T_{in} and the outlet temperature T_{out}.

When wind is present, the wind pressure on each external facade of the room superimposes an additional pressure. This implies there is a counter-balancing pressure P_{cb} from the interior of the room onto the bottom of the stack so that the flows of air into and out of the room will balance.

The pressure difference P_{wi} between the wind pressure on the wall *i* and the (static) local outdoor air pressure at the same location is given by

$$P_{wi} = \frac{1}{2} C_{pi} \rho_a v_i^2, \tag{1}$$

where C_{pi} is a local coefficient depending on the direction of the wind on the facade, ρ_a is the ambient air density, and v_i is the wind speed.

The volume flow rate v_i of air into the room through openings of area A_i in the wall is assumed to be given by the empirical formula

$$\dot{V}_i = A_i C_{Di} P_{wi}^n, \tag{2}$$

where C_{Di} is a discharge coefficient depending on the direction of the wind on the facade, and n is an exponent between 0.4 for large openings and 1.0 for small openings.

The pressure loss due to friction in the pipe P_f is given by the Darcy-Weisbach equation as [11]:

$$P_f = f \frac{h_s}{D_h} \rho_a \frac{v_{in}^2}{2}, \tag{3}$$

where f is the friction factor, D_h is the hydraulic diameter, and v_{in} is the velocity of the airflow through the stack. When wind flows past the outlet of the stack there is a wind pressure P_w at the outlet. Then the upward buoyancy pressures ΔP_{in} and ΔP_{out} at the inlet and outlet of the stack, respectively, are given by

$$\Delta P_{in} = -\frac{g \rho_a (-h_{NL})(T_s - T_a)}{T_s} - P_{cb} \tag{4}$$

and

$$\Delta P_{out} = -\frac{g \rho_a (h_s - h_{NL})(T_s - T_a)}{T_s} + P_w - P_f, \tag{5}$$

where g is the acceleration of gravity.

The volumetric rates of air flow $\dot{V}_{in}$ into and $\dot{V}_{out}$ out of the stack are determined by

$$\dot{V}_{in} = A_s C_{Di} \frac{\Delta P_{in}}{|\Delta P_{in}|^{1-n}} \text{ and } \dot{V}_{out} = A_s C_{Di} \frac{\Delta P_{out}}{|\Delta P_{out}|^{1-n}}, \tag{6}$$

where A_s is the cross-sectional area of the stack [6].

3.2 Heat transfer in the stack

The temperature of the air entering the stack at the bottom is assumed to be the ambient air temperature T_a. Let T_{wi} be the temperature of the water flowing into the top of the water jacket around the stack from the solar water heater. Then the temperature T_{out} to which the air emerging from the stack is raised is given by [7].

$$T_{out} = T_a + \varepsilon (T_{wi} - T_a), \tag{7}$$

where ε is the effectiveness of the heat exchange process in the stack. This is given by

$$\varepsilon = \frac{1 - \exp[-N(1-R)]}{1 - R\exp[-N(1-R)]}, \tag{8}$$

where N is the number of heat transfer units, and R is ratio of the heat capacity flow rates of air M_a and water M_w. These quantities are given by

$$N = \frac{UA_a}{M_a} \text{ and } R = \frac{M_a}{M_w}, \tag{9}$$

where U is the overall heat transfer coefficient and A_a is the heat transfer area of the heat exchanger. The value of U is related to the heat transfer rate Q by the equation,

$$Q = UA_a \Delta T, \tag{10}$$

where ΔT is the log mean temperature difference across the heat exchanger:

$$\Delta T = \frac{(T_{wi} - T_{out}) - (T_{wo} - T_a)}{\ln[(T_{wi} - T_{out})/(T_{wo} - T_a)]},$$

and T_{wo} is the temperature of the water flowing out the bottom of the water jacket around the stack.

4. Calculations and Experiments

4.1 Design calculations for the ventilation stack

For ventilating the experimental room it was assumed that an airflow rate half of that used in a one-ton air conditioner would be sufficient. This is half of 400 cubic feet per minute, or about 0.1 m^3s^{-1}, or about 10 air changes per hour in this experimental room. Assuming ventilation for four hours per day and a rise in temperature the air passing through the stack of 5 °C, the amount of heat required on the air side was about 2.0 kWh per day.

In order to estimate the size of the solar collector required to supply this heat it was assumed that there was no heat lost in the heat exchange process, the efficiency of the solar collector was 50% and the solar radiation on the collector was 18.2 MJm^{-2} per day, i.e. 5 $kWhm^{-2}$ per day (a typical average for sites in Thailand). This indicates that a solar collector area of 0.8 m^2 should be sufficient, but the collector available in the market and used in our experiments had an area of 2.16 m^2 and a water storage tank with capacity 160 liters.

Various stack designs were studied and the one finally adopted, as described earlier in section 2, was predicted to have the design volumetric air flow rate of 0.1 m^3s^{-1} when the hot water from the solar collector was pumped through the stack at 100 liters per hour.

4.2 Experiment on ventilation

An experiment was conducted to determine the amount of ventilation produced by the stack with solar heated water pumped through the water jacket for four hours during the daytime from 10:00 to 14:00. Measurements were made of the water flow rate, the temperature T_{wi} of the water entering the top of the stack, the temperature T_{wo} of the water from the bottom of the stack, the temperature T_{out} of the air emerging from the top of the stack and the air speed v_{in} into the bottom of the stack. Type K thermocouples were used for the temperature measurements, and a hot wire anemometer was used for measuring the air speed into the stack. (It was found by a separate check that the air speed was the same in all twelve air ducts.) Measurements were also made of the wind speed and direction, the ambient air temperature T_a, and the ambient relative humidity outside the building.

4.3 Comparison between calculated and measured airflow through the stack

The airflow through the stack was calculated using the equations in section 3.1 and 3.2 and the measurements mention in section 4.2. In this calculation, the values of h_{NL}, T_s, and P_{cb} are unknown. To start the calculation assumed values of h_{NL}, $v_{in,}$, and P_{cb} were taken. The airflow into the room $\sum \dot{V}_i$ was first calculated using the measured value of v_i and equations (1) and (2). Next T_s and the airflow $\dot{V}_{\text{out}}$ from the stack were calculated using equations (3) to (9). Then the airflow through the stack must satisfy the mass conservation conditions $\dot{V}_{\text{in}} = \sum \dot{V}_i$ and $\dot{V}_{\text{in}} = \dot{V}_{\text{out}}$, and the calculated air speed $v_{in} = \dot{V}_{\text{in}} / A_s$ should be equal to the measured air speed. The three assumed values h_{NL}, $v_{in,}$, and P_{cb} were adjusted by the Newton-Raphson iteration method until these conditions were satisfied.

4.4 Experiment on the light pipe

During daytime the stack functions as a light pipe to transmit daylight into the building interior, as shown in Fig. 4. Measurements were made of the beam, diffuse and global illuminance from the sky on the site of the experiment at a distance of 100 m from the experimental building. Measurements were also made of the illuminance at the top of the light pipe under the glass cover and the illuminance under the diffuser plate at the exit of the light pipe.

4.5 Calculation of daylight transmission through the light pipe

The calculation of the daylight transmission through the light pipe was done by using the Building Energy Simulation Program (BESim) [8] with the components and dimensions shown and Figs. 2 and 3, and the measurements mentioned above.

5. Results and discussion

5.1 Air flow through the stack

Values of the local outdoor wind pressure coefficients C_p and the discharge coefficients C_D used in equations (1) , (2) and (6) are shown in Table 1. The exponent used in equations (2) and (6) was n = 0.65. These values were taken from reference [6].

Table 1. Configuration of experimental room for ventilation calculation

Facade	South	East	West	North	stack
Inlet area(m^2)	0.1	0.1	0.1	0.1	0.23
Cp	0.54	-0.39	0.54	-0.39	-0.44
C_D	0.65	0.65	0.65	0.65	0.3

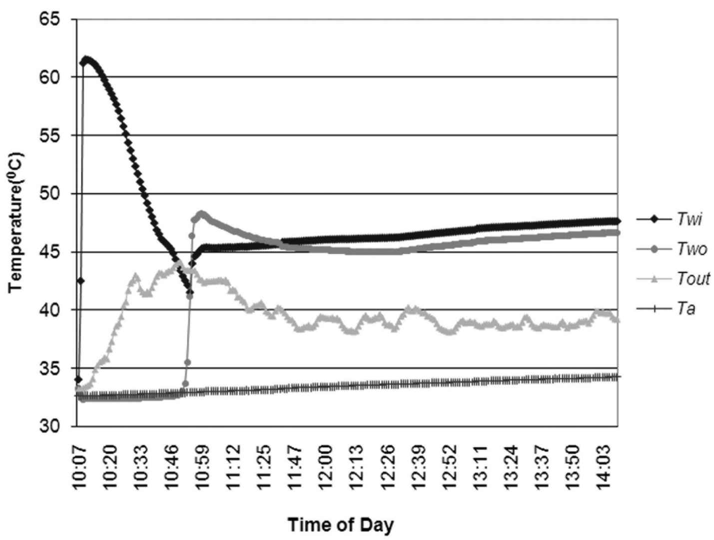

Figure. 5. Measured temperatures in the stack.

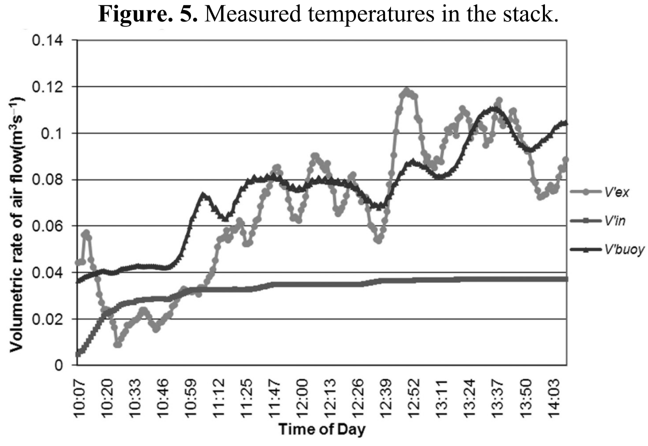

Figure. 6. Comparison between experimental and calculated volumetric airflow rates.

The measured temperatures T_a, T_{wi}, T_{wo}, and T_{out} for an experiment conducted on 1 June 2010 are shown in Fig. 5. The average equilibrium temperatures taken from the data in Fig. 5 are shown in Table 2. Figure 6 shows the measured volumetric airflow rate $\dot{V}_{ex}$ through the stack in the same experiment together with the calculated total airflow rate $\dot{V}_{in}$ and the air flow rate $\dot{V}_{buoy}$ calculated from the thermal buoyancy effect only. Table 3 shows the average volumetric airflow rates for three periods of the day taken from the data in Fig. 6. The temperatures of the two fluids given in Table 2 were used for calculating the effectiveness and the overall heat transfer coefficient of heat exchanger.

Table 2. Average temperatures at equilibrium (°C)

T_{wi}	T_{wo}	T_a	T_{out}
47	46	34	39

Table 3. Average volumetric airflow rates (m^3s^{-1}) from the data in Fig. 6

Time	$\dot{V}_{ex}$	$\dot{V}_{in}$	$\dot{V}_{buoy}$
10:07 - 11:14	0.029	0.048	0.026
11:14 - 12:30	0.071	0.077	0.034
12:31 - 14:09	0.093	0.092	0.036

Table 3 shows that in the third period of the day the experimental and calculated volumetric airflow rates $\dot{V}_{ex}$ and $\dot{V}_{in}$ respectively were almost equal to the design value 0.1 m^3s^{-1} given in section 4.1. The volumetric airflow rate $\dot{V}_{buoy}$ calculated from the thermal buoyancy effect only was approximately one third of the design value. On this particular day the wind effect was small during the first time period, and increased to approximately double the buoyancy effect during the third period. The calculated total airflow rates were in satisfactory agreement with the experimental values.

The heat exchanger effectiveness can be calculated indirectly using the temperatures in Table 2, the water and air heat capacity flow rates derived from the data in Table 3, and the heat transfer area in the stack using equations (8) to (10). For this purpose averaged values over the four hours of the experiment were used. The average hot water flow rate was 280 liters per hour. The measured average heat capacity flow rates M_w and M_a of the water and air were 326 WK^{-1} and 57 WK^{-1} respectively. The average measured heat transfer rate from the water was 326 W, and average measured heat transfer to the air was 286 W. Therefore the measured heat loss rate from the heat exchanger was 40 W. The overall heat transfer coefficient U calculated from the measured data and equation (10) was 1.463 W $m^{-2}K^{-1}$. When the average calculated volumetric rate $\dot{V}_{in}$ was used to estimate the number N of heat transfer units, the heat exchange effectiveness ε was estimated to be 0.35. When the wind effect was omitted and $\dot{V}_{buoy}$ was used, the heat exchange effectiveness ε was estimated to be 0.53. The temperatures in Table 2 and equation (7) give the heat exchanger effectiveness directly as $\varepsilon = 0.48$, which lies between these two estimated values. The measured amount of heat transferred to the air during the four hours of the experiment was 1.144 kWh compared with the design value of 2 kWh.

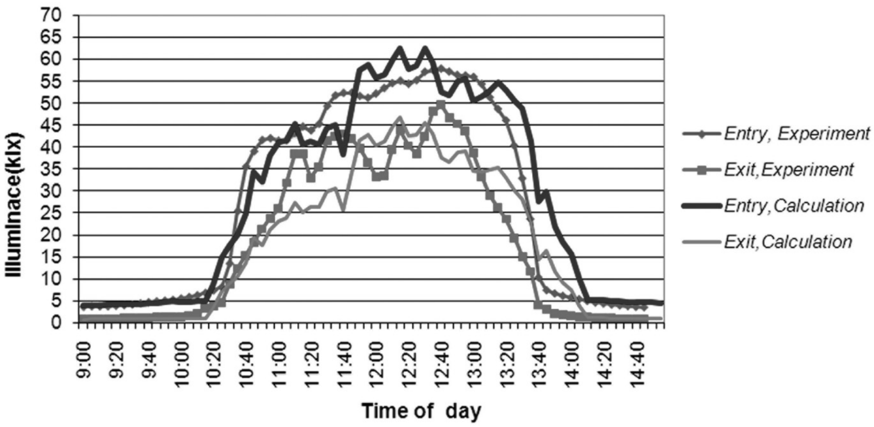

Figure. 7. Comparison between experimental and calculated illuminances at the entrance and exit of the of light pipe.

5.2 Daylight transmission through the light pipe

An experiment to measure the performance of the light pipe was conducted on 2 June 2010. The results are shown in Fig. 7. The very low illuminances before 10:20 and after 14:10 are caused by shading of the entrance to the light pipe by the slatted cover (see Fig. 4). There is fairly good agreement between the experimental and calculated values. At noon the illuminance at the entrance was 59 klx and the illuminance at the exit was 33 klx, giving a transmission of 56%, but at 12:40 the transmission was 84%. These variations were caused by varying cloudiness. The calculated values shown in Fig.7 were based on illuminances measured at the weather station on the experimental site. They correspond to a light pipe transmission of 75%.

6. Conclusions

The ventilation due to the stack effect only (ignoring the wind effect) amounted to an air change rate of 3.4 per hour. This value was lower than the design value of 10 per hour. Nevertheless, the total effect (including the wind effect) in this research gave ample ventilation in this experimental room.

The transmission of illuminance through our light pipe in the middle of the partly cloudy day varied in the approximate range 56-84%.

A quantitative comparison between our results and those of Oakley, et al. [1] cannot be made due to the widely different conditions in two cases. However, the results are qualitatively similar.

Our research has demonstrated the usefulness of solar heating to enhance ventilation through a stack that is also used as a light pipe, and we have shown that the mathematical model used could be developed as a design tool.

Acknowledgements

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